

Integration of a free-piston Stirling engine and a moving grate incinerator

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Abstract

The feasibility of recovering the waste heat from a small-scale incinerator (designed by Industrial Technology Research Institute) and generating electric power by a linear free-piston Stirling engine is investigated in this study. A heat-transfer model is used to simulate the integration system of the Stirling engine and the incinerator. In this model, the external irreversibility is modeled by the finite temperature difference and by the actual heat transfer area, while the internal irreversibility is considered by an internal heat leakage. At a fixed source temperature and a fixed sink temperature, the optimal engine performance can be obtained by the method of Lagrange multipliers.

From the energy and mass balances for the interesting incinerator with the feeding rate at 16 t/d, there is enough otherwise wasted energy for powering the Stirling engine and generate more than 50 kW of electricity.

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1. Introduction

Incineration with heat recovery is an attractive process of recovering energy in some countries [1,2]. The conversion of waste heat to power is very desirable to the areas without sufficient energy resources. It has been a common practice to convert waste into superheated steam by absorbing the otherwise wasted energy, the obtained steam can then be used to power a steam turbine and generate electricity (called steam turbine GENSET). For instance, in Japan 769 MW of electric power was generated by 181 large-scale incineration plants in 1998 [3]. However, it is not appropriate to use steam turbine GENSET to recover energy from a small-scale incinerator (say less than 500 t/d). This is because the thermal efficiency of steam turbine will drastically drop as the turbine capacity decreases [4,5]. On the other hand, Stirling engine can retain high efficiency over power range from a few tens of watts to several

kilowatts and is very suitable to use as Stirling GENSET for a small or medium scale incinerator.

Among several types of Stirling machines, the free-piston Stirling engine matched with a linear alternator (called free-piston Stirling GENSET) has the most mechanical simplicity and the least friction loss due to mechanical movement [6]. Some technical difficulties inherent to linear alternator such as the piston's overstroking and underdamped problems have been gradually resolved [7]. With many great features in power generation, the free-piston Stirling GENSET has been considered in this study as the potential candidate to recover the waste energy from a small-scale incinerator, designed by the Industrial Technology and Research Institute (ITRI) of Taiwan.

Since the Stirling engine is an external combustion engine, which means the energy used to power the engine movement is entirely obtained from the external source, the losses due to the heat transfer in and out of engine or due to the heat bypass within the engine structure may be more important than that of mechanical losses. Recently, Kongtragool and Wongwises [8,9] performed the studies and found the power and efficiency of a Stirling engine were strongly affected by the irreversible heat transfer process

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Nomenclature

$A_{in,H}$	the inner area of engine's hot-end surface (m^2)	$\dot{Q}_{source}$	heat transfer from the heat source (W)
$A_{out,C}$	the outer area of engine's cold-end surface (m^2)	$\dot{Q}_T$	thermal leakage and other internal irreversibilities (W)
$A_{out,H}$	the outer area of engine's hot-end surface (m^2)	T_C	the surface temperature of engine's cold-end (K)
$F_{H,O}$	the view factor from hot-end surface to source surface	T_{fgas}	the temperature of flame gases (K)
$h_{coolant}$	the convective coefficient of cooling water ($W/m^2/K$)	T_H	the surface temperature of engine's hot-end (K)
h_{He}	the convective coefficient of helium gas ($W/m^2/K$)	T_L	lowest fluid temperature (K)
$(HA)_C$	the thermal conductance from the inner fluid to the outer cold-end surface (W/K)	T_{sink}	the temperature of heat sink (K)
$(HA)_{fgas}$	the thermal conductance between hot-end surface and flame gases (W/K)	T_{source}	the temperature of heat source (K)
$(HA)_H$	the thermal conductance from the outer hot-end surface to the inner fluid (W/K)	T_U	highest fluid temperature (K)
$(HA)_{sink}$	the thermal conductance from the outer cold-end surface to the heat sink (W/K)	$\dot{W}$	cycle-average power (W)
$\dot{Q}_C$	the heat transfer from working fluid to engine's cold-end surface (W)	α	absorptivity of the surface exposed to flame
$\dot{Q}_{conv}$	convection heat transfer from heat source to hot-end of engine (W)	δ	$(HA)_{sink}/(HA)_C$
$\dot{Q}_H$	heat transfer from hot-end surface to working fluid (W)	ε	$(HA)_C/(HA)_H$
$\dot{Q}_{rad}$	radiation heat transfer from heat source to hot-end of engine (W)	η_{Carnot}	the thermal efficiency of Carnot cycle
$\dot{Q}_{sink}$	the rejected heat transfer from working fluid (W)	η_{endo}	the thermal efficiency of internal reversible cycle
		η	thermal efficiency
		θ	$(HA)_{sink}/(HA)_H$
		λ_m	eigen-values of Lagrange Multiplier equations
		ξ	the ratio of heat which actually flow into the fluid
		σ	Stefan–Boltzmann constant, $5.67 \times 10^{-8} W/m^2 K^4$
		τ	$\alpha A_{H,O} F_{H,O} \sigma / (HA)_H$
		ϕ_m	constraint function m

[8], and also affected by regenerator effectiveness and dead volume [9].

In this study, a heat transfer model is used to simulate a realistic Stirling engine, the effects due to engine's irreversibilities on the output power and thermal efficiency will be modeled by mathematical formulation, and Lagrange multipliers method is employed to optimize the net-work output under the fixed source and sink temperatures.

2. Incinerator and waste heat

The designed incinerator is of a modular type composed of a feeding gear, a dual combustor with primary and secondary combustion chambers, a heat recovery system, and an air pollution control system. The burning capacity ranges 16–50 t/d.

In order to comply to the regulation set by the Environmental Protection Administration of Taiwan to eliminate toxic gases, the temperature of secondary chamber has to be kept above 1273 K. Therefore, the Stirling GENSET is intended to be installed on the boundary of secondary combustor with the hot-end surface embedded inside the chamber, see Fig. 1. The control of chamber's temperature is automatically made by adjusting

an auxiliary fuel pump, the flow rate of preheated air from the heat recovery system, and the air flow rate directly from outside ambient.

The available waste heat at the feeding of 16 t/d was estimated to be 314 kW from the energy balance for both combustors [10]. The estimation was performed under the conditions of burning the sample waste with 100% excess air. A complete combustion was assumed, lower heating values were used in the calculation, and no auxiliary fuel was burnt. The content data for sample waste was provided by ITRI, in which they have analyzed the buck wastes collecting from four different industrial parks.

3. Heat transfer model

To recover the waste heat, the hot-end surface of Stirling engine is embedded inside the combustion chamber, see Fig. 1. The temperature of heat source T_{source} , shown in Fig. 2, represents the controlled chamber temperature of incinerator. The temperature of heat sink T_{sink} in Fig. 2 stands for the temperature of coolant used to cool down the cold-end surface, see Fig. 1. T_H is the surface temperature of engine's hot end (receiving flame heat), and T_C the surface temperature of engine's cold end (contacting with coolant). Since the power cycle (thermodynamic cycle) is

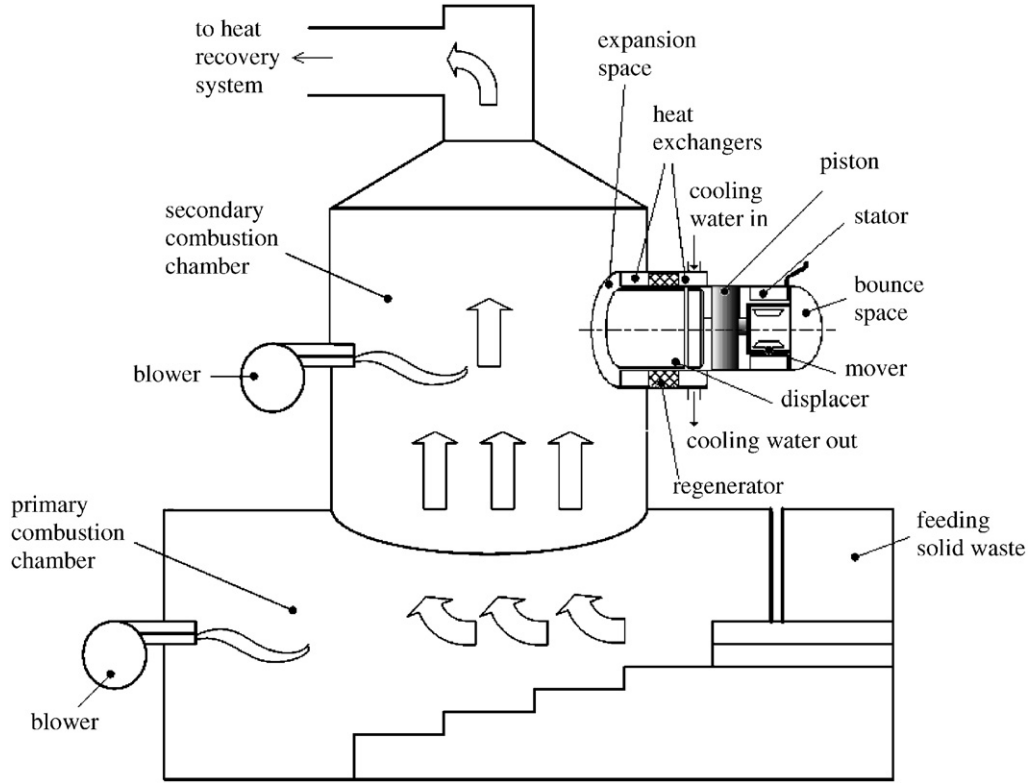


Fig. 1. Schematic diagram of the integration system.

completed by the heating, expansion, cooling, contraction processes of working fluid, the highest fluid temperature (the fluid receiving hot-end heat), T_U , is lower than T_H and the lowest fluid temperature (the fluid adjacent to cold-end surface), T_L , is higher than T_C . The magnitude sequence of these temperatures is thus to be $T_{\text{source}} > T_H > T_U > T_L > T_C > T_{\text{sink}}$.

3.1. Heat transfer to engine

The cycle-average heat transfer from the heat source to the engine's hot end can be expressed as the sum of radiation heat transfer and convection heat transfer; i.e.

$$\dot{Q}_{\text{source}} = \dot{Q}_{\text{rad}} + \dot{Q}_{\text{conv}}, \quad (1)$$

where

$$\dot{Q}_{\text{rad}} = \alpha A_{\text{out,H}} F_{H,O} \sigma (T_{\text{source}}^4 - T_H^4) \quad (2)$$

and

$$\dot{Q}_{\text{conv}} = (HA)_{\text{fgas}} (T_{\text{fgas}} - T_H). \quad (3)$$

In Eq. (2), α is the absorptivity of the surface exposed to flame and σ is the Stefan–Boltzmann constant, which is $5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$. It should be noted that the reciprocity relation for view factors ($A_1 F_{12} = A_2 F_{21}$) has been applied in Eq. (2), therefore, $A_{\text{out,H}}$ is the external area of engine's hot-end and $F_{H,O}$ is the view factor from hot-end surface to source surface. In Eq. (3), $(HA)_{\text{fgas}}$ is the product of area $A_{H,0}$ and the convective heat transfer coefficient between hot-end surface and flame gases. T_{fgas} is the

temperature of flame gases, we assume $T_{\text{fgas}} = T_{\text{source}}$ for simplicity purpose.

3.2. Heat transfer to working fluid

The cycle-average heat transfer from hot-end surface to working fluid is

$$\dot{Q}_H = (HA)_H (T_H - T_U). \quad (4)$$

$(HA)_H$ is the reciprocal of the total thermal resistance from the outer hot-end surface to the inner fluid. The thermal resistance of a conducting metal (usually Inconel alloy) is usually much lesser than the convection resistance of the working fluid (usually helium gas). $(HA)_H$ can be simplified to $h_{\text{He}} A_{\text{in,H}}$, where h_{He} is the convective coefficient of helium gas and $A_{\text{in,H}}$ is usually fabricated into a fin shape (microfin, etc.) such that $A_{\text{in,H}}$ is about 3~7 times of $A_{\text{out,H}}$.

3.3. Heat rejection from working fluid

The cycle-average heat transfer from working fluid to engine's cold-end surface is

$$\dot{Q}_C = (HA)_C (T_L - T_C), \quad (5)$$

similar to $(HA)_H$, $(HA)_C \cong h_{\text{He}} A_{\text{in,C}}$ where h_{He} is the convective heat transfer coefficient of working fluid (helium gas), and $A_{\text{in,C}}$ is the inner area of engine's cold-end surface.

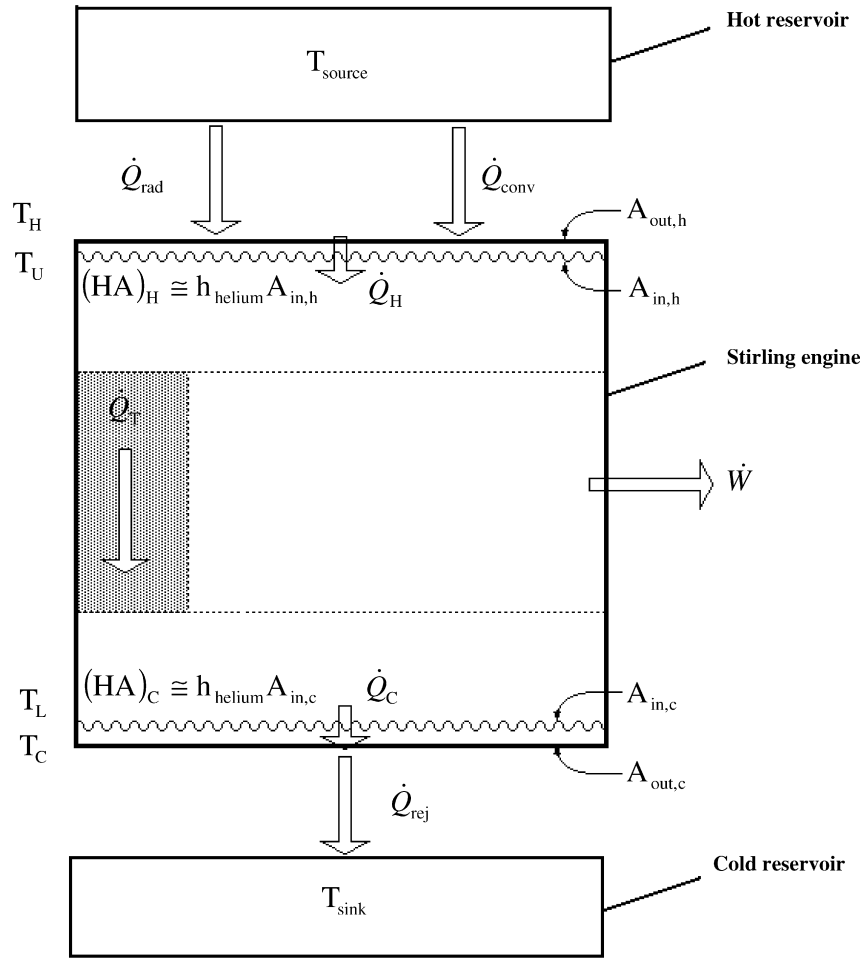


Fig. 2. Heat transfer model.

The rejected heat from working fluid $\dot{Q}_C$ is eventually carried away by the cooling water, which is

$$\dot{Q}_{\text{sink}} = (HA)_{\text{sink}}(T_L - T_{\text{sink}}). \quad (6)$$

Again, $(HA)_{\text{sink}} \cong A_{\text{out,C}} h_{\text{coolant}}$ where $A_{\text{out,C}}$ is the outer area of engine's cold-end surface and h_{coolant} is the convective heat transfer coefficient of cooling water.

3.4. Thermal leakage

Inside a typical Stirling engine, the temperature of hot-end surface (constantly kept above T_U) is much higher than that of cold-end surface (constantly staying below T_L). The heat leaking through engine structure (including regenerator and cylinder wall) without actually flowing into the working fluid to contribute to the power production is inevitable. If we express this kind of thermal leakage and other internal irreversibilities (such as friction heat etc.) as $\dot{Q}_T$, see Fig. 2.

The cycle-average power, $\dot{W}$, can be written as

$$\begin{aligned} \dot{W} &= (\dot{Q}_H - \dot{Q}_T) - (\dot{Q}_C - \dot{Q}_T) \\ &= (\dot{Q}_H - \dot{Q}_C) \\ &= (HA)_H(T_H - T_U) - (HA)_C(T_L - T_C). \end{aligned} \quad (7)$$

The thermal efficiency for a heat engine operating between heat source at T_{source} and heat sink at T_{sink} , the thermal efficiency of Carnot cycle (all processes are reversible) is

$$\eta_{\text{Carnot}} = 1 - \frac{T_{\text{sink}}}{T_{\text{source}}}. \quad (8)$$

If the reality that temperature difference must exist for the heat flow from source to working fluid or from working fluid to sink are considered the thermal efficiency should be written as

$$\eta_{\text{endo}} = \frac{\dot{W}}{\dot{Q}_H} = 1 - \frac{T_L}{T_U}, \quad (9)$$

where η_{endo} represents the thermal efficiency of internal reversible (or endor-reversible) cycle, and T_L and T_U are the lowest and highest temperatures of engine's working fluid. In reality $T_{\text{source}} > T_U$ and $T_L > T_{\text{sink}}$, all the heat transfer processes are irreversible, and because the energy transfers are conducted across the engine's boundary, these processes are called external irreversibilities.

In addition, when the internal irreversibility is considered as the internal heat-leakage $\dot{Q}_T$, the amount of heat actually flow into the working fluid (i.e. actually contribute

the power generation) is $(\dot{Q}_H - \dot{Q}_T)$, we can express

$$\frac{\dot{W}}{\dot{Q}_H - \dot{Q}_T} = 1 - \frac{T_L}{T_U}. \quad (10)$$

Here, $\dot{Q}_T$ is used to represent the overall internal irreversibility. If we define the ratio of heat which actually flow into the fluid as ξ , i.e.,

$$\xi = \frac{\dot{Q}_H - \dot{Q}_T}{\dot{Q}_H},$$

the thermal efficiency of a more realistic engine, Eq. (10), can be rewritten as

$$\eta = \frac{\dot{W}}{\dot{Q}_H} = \xi \left(1 - \frac{T_L}{T_U} \right). \quad (11)$$

4. Optimization (Lagrange multiplier method)

From designer's point of view, it is interesting to know what kind of temperature combination will result in the best performance if the engine is operating under the fixed source and sink temperatures.

In order to obtain the best performance, the work done and represented by Eq. (7) is considered as the objective function to be maximized. Eq. (7) is rewritten as the *objective function* of

$$\dot{W} = (HA)_H(T_H - T_U - \varepsilon T_L + \varepsilon T_C), \quad (12)$$

where $\varepsilon = (HA)_C/(HA)_H$

Since the maximum power is limited by Eq. (11) for an actual engine, Eq. (11) is rewritten and used as the first constrained function:

$$\frac{(HA)_H(T_H - T_U - \varepsilon T_L + \varepsilon T_C)}{(HA)_H(T_H - T_U)} = \xi \left(1 - \frac{T_L}{T_U} \right),$$

or the *constraint function 1*:

$$T_C - T_L + \frac{1}{\varepsilon}(T_H - T_U) - \frac{\xi}{\varepsilon} \frac{(T_U - T_L)(T_H - T_U)}{T_U} = 0. \quad (13)$$

In addition, the power of engine is closely related to the heat flows in and the heat flows out of engine. The second constraint is $\dot{Q}_H = \dot{Q}_{\text{source}}$, from Eqs. (1)–(4).

$$\alpha A_{H,O} F_{H,O} \sigma (T_{\text{source}}^4 - T_H^4) + (HA)_{HS}(T_{\text{source}} - T_H) = (HA)_H(T_H - T_U),$$

or the *constraint function 2*:

$$\tau T_{\text{source}}^4 - \tau T_H^4 + \theta T_{\text{source}} - \theta T_H - T_H + T_U = 0, \quad (14)$$

where $\tau = \alpha A_{H,O} F_{H,O} \sigma / (HA)_H$, and $\theta = (HA)_{\text{sink}} / (HA)_H$.

The third constraint is $\dot{Q}_C = \dot{Q}_{\text{sink}}$, from Eqs. (4) and (5).

$$(HA)_C(T_L - T_C) = (HA)_{\text{sink}}(T_C - T_{\text{sink}}),$$

or the *constraint function 3*:

$$T_L = T_C + \delta T_C - \delta T_{\text{sink}}, \quad (15)$$

where $\delta = (HA)_{\text{sink}} / (HA)_C$.

If T_{source} and T_{sink} are fixed, and the thermal conductances are set to be some practical values, the best combination of T_H , T_U , T_L , and T_C can be obtained by solving the Lagrange multiplier equations, which are:

$$\nabla \dot{W} - \sum_{m=1}^3 \lambda_m \nabla \phi_m = 0, \quad (16)$$

$$\begin{aligned} \phi_1(T_H, T_U, T_L, T_C) &= T_C - T_L + \frac{1}{\varepsilon}(T_H - T_U) \\ &\quad - \frac{\xi}{\varepsilon} \left(T_H - T_U - \frac{T_H T_L}{T_U} + T_L \right) = 0, \end{aligned} \quad (17)$$

$$\begin{aligned} \phi_2(T_H, T_U, T_L, T_C) &= \tau T_{\text{source}}^4 - \tau T_H^4 + \theta T_{\text{source}} \\ &\quad - \theta T_H - T_H + T_U = 0, \end{aligned} \quad (18)$$

$$\phi_3(T_H, T_U, T_L, T_C) = T_L - T_C - \delta T_C + \delta T_{\text{sink}} = 0. \quad (19)$$

The above constraint functions ϕ_1 , ϕ_2 , and ϕ_3 , are essentially the same as Eqs. (13)–(15). Since there are four independent variables (T_H , T_U , T_L , T_C), the gradient $\dot{W}$ and gradient ϕ_m shown in Eq. (16) are the derivative of objective function and constraint functions to each independent variable, which will result in four equations. For example if one takes the partial derivative of Eq. (16) with respect to T_H , $\partial(16)/\partial T_H$, he gets

$$\begin{aligned} (HA)_H - \frac{1}{\varepsilon} \lambda_1 \left(1 - \xi + \xi \frac{T_L}{T_H} \right) \\ + 4\tau \lambda_2 T_H^3 + \theta \lambda_2 + \lambda_2 = 0. \end{aligned} \quad (20)$$

Similarly, from $\partial(16)/\partial T_U$, $\partial(16)/\partial T_L$, and $\partial(16)/\partial T_C$ one will, respectively, obtain the other three equations:

$$-(HA)_H + \lambda_1 \left(\frac{1}{\varepsilon} - \frac{\xi}{\varepsilon} + \frac{\xi T_H T_L}{\varepsilon T_U^2} \right) - \lambda_2 = 0, \quad (21)$$

$$-\varepsilon (HA)_H + \lambda_1 \left(1 + \frac{\xi}{\varepsilon} - \frac{\xi T_H}{\varepsilon T_U} \right) - \lambda_3 = 0, \quad (22)$$

$$\varepsilon (HA)_H - \lambda_1 + \lambda_3 + \lambda_3 \delta = 0. \quad (23)$$

Equations (17)–(23) can be solved simultaneously to obtain the optimized value of T_H , T_U , T_L , T_C and the eigen-values λ_1 – λ_3 . The power output, thermal efficiency and other heat transfer quantities can then be calculated.

5. Results and discussion

It is interesting to know how much of power output can be obtained from a properly designed Stirling engine with an unit outer hot-end surface, i.e. we set $A_{\text{out},H} = 1 \text{ m}^2$. Since $(HA)_H \cong h_{\text{He}} A_{\text{in},H}$, the convective heat transfer coefficient of helium gas is about twice of flue gases, and

the inner finned surfaces at both hot and cold ends are assumed five times of the outer surfaces. It is reasonable to set $(HA)_H = 1000 \text{ W/K}$, $(HA)_{\text{fgas}} = 100 \text{ W/K}$, $T_{\text{source}} = 1273 \text{ K}$, and $T_{\text{sink}} = 300 \text{ K}$.

For an endor-reversible heat engine operating steadily between two fixed temperatures, Bejan [11] found the engine's best performance can be obtained if the total thermal conductance (HA) is evenly divided for hot-end and cold-end heat exchangers, i.e. $(HA)_H = (HA)_C$. In our previous study [12], we derived a more general (HA) formulation for a realistic engine. Under a fixed total thermal conductance constraint, $(HA)_H$ is smaller but very close to $(HA)_C$ if the internal irreversibility is considered.

From the above reasonings, we set $\varepsilon = (HA)_C/(HA)_H = 1$, and $\delta = (HA)_{\text{sink}}/(HA)_C = 1$. When $\dot{Q}_T/\dot{Q}_H = 0.3$ (or $\xi = 0.7$), the solutions of Eqs. (17)–(23) are $T_H = 995 \text{ K}$, $T_U = 887.9 \text{ K}$, $T_L = 438.3 \text{ K}$, $T_C = 369.2 \text{ K}$, $\lambda_1 = 1203$, $\lambda_2 = -172.1$, and $\lambda_3 = 101.6$. The resulting heat transfers of $\dot{Q}_{\text{source}} = 107,100 \text{ W}$, $\dot{Q}_T = 32,130 \text{ W}$, $\dot{Q}_{\text{sink}} = 69,200 \text{ W}$, network $\dot{W} = 37,969 \text{ W}$, and thermal efficiency $\eta = 0.3544$. Fig. 3 shows the calculated $\dot{Q}_{\text{source}}$, $\dot{W}$ and η when the degree of heat bypass is changed, and Fig. 4 depicts the associated temperature distributions when heat bypass is varied. At $\xi = 1.0$ (no heat leakage), the calculated thermal efficiency = 53.33%, which is close to the Curzon–Ahlborn efficiency of $\eta_{\text{CA}} = 1 - (T_{\text{sink}}/T_{\text{source}})^{0.5} = 51.45\%$. As expected, the thermal efficiency as well as the output power decrease when the degree of leakage increases (ξ decreases). At $\xi = 0.75$ (25% heat bypass), the calculated thermal efficiency is about 38.2% which is close to 50% of the Carnot efficiency = $1 - (T_{\text{sink}}/T_{\text{source}}) = 76.43\%$. Since the optimal temperatures are slightly increased when the degree of heat bypass increases (ξ decreases), see Fig. 5, the change rate of efficiency is not exactly proportional to the change rate of heat bypass.

If the source and sink temperature are still kept at 1273 and 300 K, but the value of thermal conductance, $(HA)_H$, is changed, and the ratios of thermal conduction are kept

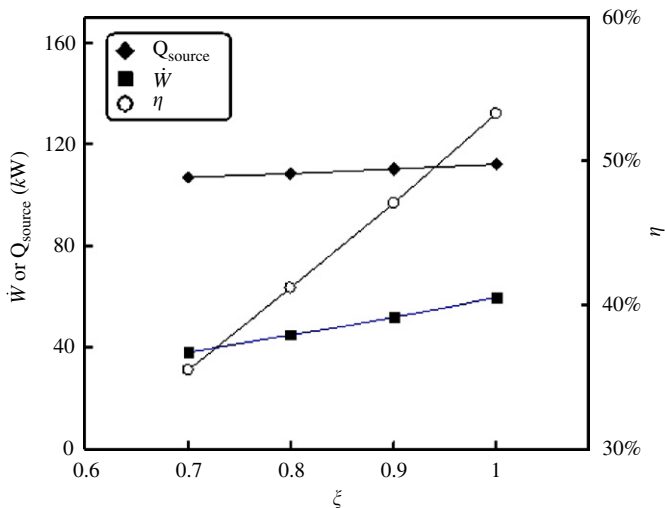


Fig. 3. Heat input, network and efficiency at different degrees of heat loss.

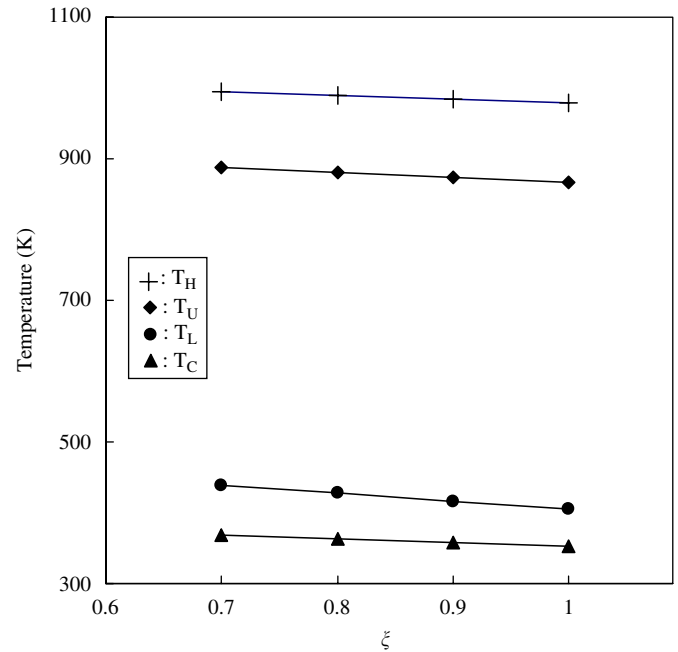


Fig. 4. T_H , T_U , T_L , and T_C values at different degrees of heat loss.

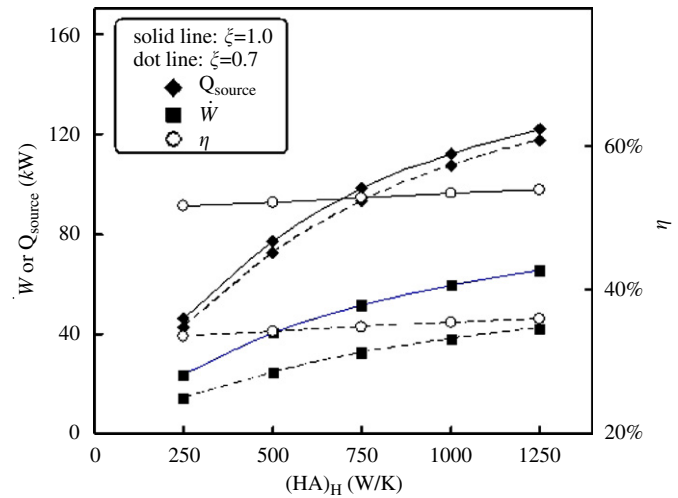


Fig. 5. Heat input, network and efficiency at different $(HA)_H$.

unchanged, the calculated $\dot{Q}_{\text{source}}$, $\dot{W}$ and η are presented in Fig. 5 and the temperature distributions are presented in Fig. 6. Results indicate $\dot{Q}_{\text{source}}$, $\dot{W}$ and η are increasing and all of the temperatures are decreasing as the thermal conductance is getting larger. Since T_H represents the highest temperature of engine body, the predicted value of T_H provides important data to select the hot-end material. Fig. 6 indicates the curve of T_H has the largest decreasing rate. Since temperature is the easiest data to measure, from the designer's point of view, all of the calculated optimal temperatures can be used as reference values to compare with the measured data during the development stage. This kind of comparisons can provide some insight information to perform structural adjustments or local fine turnings to

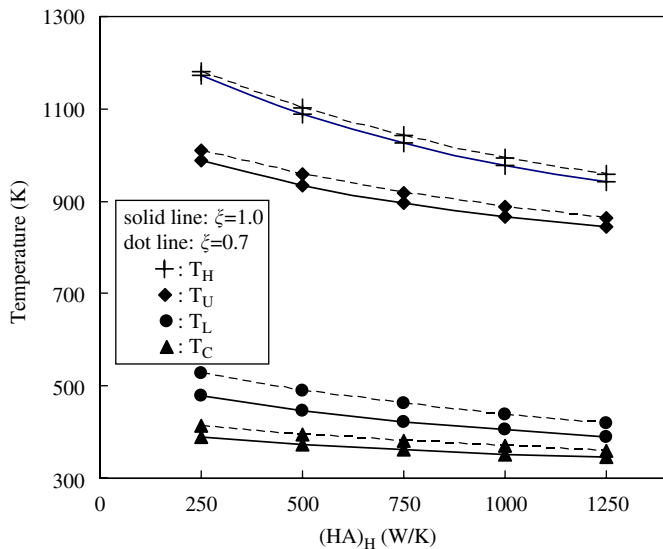


Fig. 6. T_H , T_U , T_L , and T_C values at different $(HA)_H$.

improve the overall design. From Fig. 5, we can see even at $(HA)_H = 500 \text{ W/K}$ and $\xi = 0.7$ (30% bypass heat), a very conservative estimation, the net-work (power output) is about 26 kW, and the required input heat is about 75 kW. Consider the physical dimensions of secondary combustor, there is enough space to install 2 units of Stirling GENSET, and there are more than enough waste heat to provide the engine's input heat even at the feeding rate of 16 t/d.

6. Conclusion

A simple heat transfer model is used to model the inherited irreversibilities of a realistic Stirling engine. The heat transfers from heat source to engine's hot end, and from engine's cold end to heat sink are used to represent the external irreversibilities, and the degree of heat bypass is used to simulate the overall internal irreversibility. The calculated thermal efficiency without internal bypass heat compares well with the Curzon–Ahlborn efficiency. At 25% bypass heat, the calculated thermal efficiency is very close to one-half of the Carnot efficiency.

With 1 m^2 of hot-end surface embedded inside the secondary combustion chamber, a well designed Stirling GENEST should be able to generate more than 25 kW of

power. Considering the configuration constraints of the secondary chamber and the stability of operating temperature, more than 50 kW of power can be recovered from the interesting incinerator.

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